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BOSSLER COUPLING EXPERIMENTAL FLIGHT TEST

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FINAL REPORT

(1 July 1970 to 15 February 1972)
MARCH 1972

By
R. B. Bossler, Jr.

Prepared Under Contract N00156-69-C-1316 for

NAVAL AIR PROPULSION TEST CENTER
NAVAL AIR SYSTEMS COMMAND
DEPARTMENT OF THE NAVY

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13. ABSTRACT

The Bossler coupling is a new flexible driveshaft coupling requiring experimental research to provide guidance for future design. Reported herein are bench tests of rotating driveshaft assemblies using Bossler couplings under combined speed, torque and misalignment, followed by an experimental flight test program in an HH-2D helicopter. This work is a continuation of an earlier program of endurance life testing performed under the same contract.

The test driveshaft was designed to meet or exceed the capabilities of the existing main driveshaft on the Navy/Kaman HH-2D helicopter. Ten different couplings on two different driveshaft assemblies were bench tested under combined speed, torque and misalignment. The ten couplings accumulated 196 million cycles of testing in 528 hours of rig operation. The flight test program used a Navy/Kaman HH-2D helicopter, BuNo 152191. The flight envelope was flown with the test driveshaft installed at 11,200 and 12,600 pounds gross weight. Total test driveshaft flight time was 12.5 hours as per contract. No questionable driveshaft behavior was observed. Continued flight test up to 50 hours was approved. Two concepts are described to increase coupling capability further.

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SUMMARY

The Bossler coupling is a new flexible driveshaft coupling requiring experimental research to provide guidance for future design. Reported herein are bench tests of rotating driveshaft assemblies using Bossler couplings under combined speed, torque and misalignment, followed by an experimental flight test program in an HH-2D helicopter. This work was performed under NAEC Contract N00156-69-C-1316 (Reference 1). This work is a continuation of an earlier program of endurance life testing performed under the same contract.

The test driveshaft was designed to meet or exceed the capabilities of the existing HH-2D main driveshaft, with the additional objectives of increasing reliability, providing longer life and eliminating periodic maintenance. Full-scale driveshaft assemblies were tested under combined speed, torque and misalignment. All tests were run at 6200 rpm and 18,000 pound-inches of torque. Misalignment was increased to levels which produced two coupling failures and nine coupling survivals. Ten different couplings on two different driveshaft assemblies were tested prior to release of the design for flight evaluation. The ten couplings accumulated 196 million cycles of testing in 528 hours of rig operation. These tests showed the driveshaft design to be flightworthy.

An important feature of the coupling is the ease with which it lends itself to fail-safe design, which the test driveshaft incorporated. After each bench test failure, the fail-safe structure retained all parts, allowed torque transmission to continue (somewhat reduced by the failure), and signalled the failure by an increase in vibration level.

The flight test program used a Navy/Kaman HH-2D helicopter, BuNo 152191. The HH-2D flight envelope was flown, with the aircraft at 11,200 pounds gross weight and the existing driveshaft installed, and with the test driveshaft installed. Four hours were flown at an overload gross weight of 12,600 pounds. The same flight envelope was flown, except that maximum speed was reduced by the weight increase. No questionable driveshaft behavior was observed. The total flight test elapsed time using the test driveshaft was 12.5 hours. Subsequent test driveshaft disassembly inspection and engineering review indicated that the test specimen is fully capable of further use.

It was concluded from the bench and flight tests that up to fifty hours flight testing under engineering surveillance and control could be accomplished safely and to useful purpose using the test driveshaft. Permission for this continued flight test was requested by Kaman and granted by the Naval Air Systems Command. Additionally, it was concluded that coupling capability could be increased beyond that demonstrated to date by two different design approaches. The first concept is to use variable thickness plates so that bending stress is reduced where stress concentration factors are present. The second concept is to design to operate above the rigid body resonant speeds, using the fail-safe to limit displacements during transition, in order to provide a high-speed high-angle coupling. It is recommended that flight test be continued and the concepts leading to greater capability be pursued.

FOREWORD

The Naval Air Propulsion Test Center has undertaken supervision of research programs aimed at advancing power transmission technology. Flexible driveshaft couplings have been identified as a fruitful and important area for research. The Bossler coupling is a new flexible coupling which has been investigated under Naval Air Engineering Contract N00156-69-C-1316, with technical direction provided by Mr. James D. Conboy, Aerospace Engineer, of the Naval Air Propulsion Test Center, Code AEF-2, Naval Base, Philadelphia, Pennsylvania.

This report covers bench tests of rotating driveshaft assemblies using Bossler couplings under combined speed, torque and misalignment, followed by an experimental flight test program based on the results of the bench tests. The test work reported herein was accomplished during the period from 1 July 1970 to 15 February 1972. The test work was conducted at Kaman Aerospace Corporation, Bloomfield, Connecticut, under the technical supervision of Mr. Robert B. Bossler, Jr., Chief of Mechanical Systems Research.

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Driveshaft Couplings

Driveshaft couplings accommodate the inevitable misalignments between rotating shafts in a drive train. The misalignments are caused by imperfect parts, temperature changes, and deflections of the supporting structures. The couplings accommodate these misalignments by either moving contact or flexing.

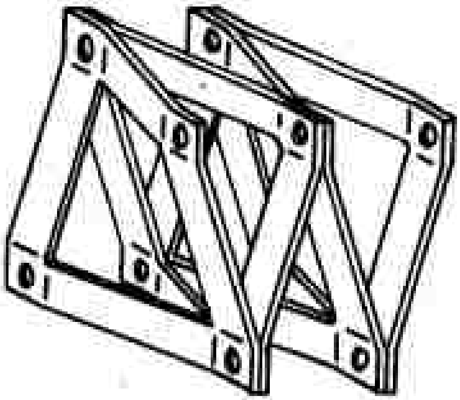
Coupling parts with moving contact require lubrication and maintenance. The rubbing parts absorb power. The lubricant and the seals limit coupling environment and coupling life. The parts wear out. The coupling may develop a large resistance to movement as the parts deteriorate. The coupling may not have constant velocity. The location of parts will be inexact because of the internal clearances required to allow motion. However, in spite of these drawbacks, many successful applications of couplings with moving contact are known. These couplings are in wide use. Coupling behavior is predictable, usually from past experience.

Couplings which accommodate misalignments by flexing avoid all the drawbacks that come with moving contact. Flexible coupling behavior, however, is not without design problems. Any flexible coupling can be proportioned with strong, thick, stiff members that easily transmit a design torque and that easily provide the stiffness to operate at a design speed. However, misalignment requires flexing of these members. The flexing produces alternating stresses that can limit coupling life. The greater the strength and stiffness of a member, the higher the alternating stress from a given misalignment. Therefore, strength and stiffness provisions to transmit torque at speed will be detrimental to misalignment capability. The problem of design is to proportion the flexible coupling to accomplish torque transmission and misalignment for the lowest system cost.

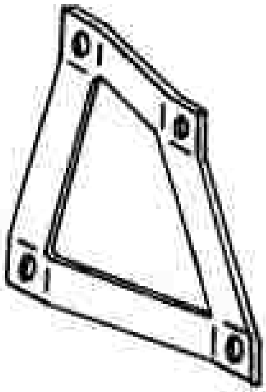
Bossler Coupling*

The Bossler coupling is a new flexible driveshaft coupling. It transmits torque through axially-loaded straight elements. It is structurally very efficient. Deformations resulting from shaft misalignments and changes in length are distributed among the many slender straight elements that comprise the coupling. The coupling geometry is illustrated on the following page.

* U.S. Patent No. 3,177,684



BOSSLER COUPLING



PLATE

ELEMENT

Mechanical characteristics of the Bossler coupling were investigated under NASA Contract NASw-1554 (Reference 2). The Abstract from the report (which is also the Summary) is presented below:

<u>Abstract (From Reference 2)</u>

"The Bossler coupling is a new, flexible, driveshaft coupling which is suitable for any level of power transmission. This report presents the results of an investigation of the mechanical characteristics of Bossler couplings. The investigation included analyses and tests. Simplified coupling analysis methods, parameter studies, and design guidelines were developed. The test areas included torsion, stiffness, strain, fatigue, constancy of velocity, critical speed, and balancing. The tests substantiated the analyses for predicting internal forces and moments, steady and alternating stress, bending and change-of-length stiffness, critical speeds, and the effects of unbalance. Experience is limited for fatigue (two tests), and for ultimate torque (six tests).

Important characteristics of the Bossler coupling were established. Velocity can be constant. The coupling has unusual capability for accommodating combined axial motion, misalignment, and torque. The coupling can survive shock-torque greatly in excess of ultimate

continuous torque and transient misalignments over three times the design continuous operating angle. Configuration modification may improve performance further. Fail-safe design is accomplished easily. The coupling has unusual characteristics that are potentially useful. The coupling appears well suited for applications requiring very long life with high reliability, very low weight, no maintenance or lubrication, and survival in hostile environments."

Endurance Life Testing

NAEC Contract N00156-69-C-1316 was awarded to Kaman Aerospace Corporation in order to provide for further Bossler coupling research. The contract was directed originally toward endurance life testing, which was performed between the period 28 February 1969 to 30 June 1970. The final report on endurance life testing, Kaman Report No. R-885, was published in August 1970 (Reference 3). The endurance life testing report is summarized briefly immediately below. It was concluded that the endurance life testing justified an experimental flight test program, which became a follow-on phase of the basic contract. The experimental flight test program is the subject of the present report.

The endurance life testing was directed toward those areas where experience was limited, as noted in the NASA Abstract previously quoted, specifically to conduct more fatigue tests and an ultimate torque test on a test coupling. The testing was further directed toward developing and demonstrating protection from a potential hazard, fretting damage. The testing evaluated various methods of fretting protection and various details of joint construction. Long endurance life of a test coupling design was substantiated by test.

An investigation of fretting protection and details of joint construction was performed on a Krouse Test Machine. Krouse testing is an economical method of screening candidate ideas. Approximately 100 million cycles of testing were used to compare fatigue-plus-wear-behavior of twenty-five test specimens. Fatigue-plus-wear life was increased from limited life at the initial test level of alternating stress to infinite life with an alternating stress 2-1/2 times the initial alternating stress. The lessons learned were applied to couplings which were then endurance life tested.

The coupling endurance life testing was performed on a non-rotating test rig. The rig applied steady torque and alternating bending to the test coupling, and induced fretting at the joints of the test coupling. Sixteen coupling tests were run. Approximately 180 million cycles of testing were used to accomplish four step-tests to failure plus one step-test stopped without failure. Success in suppressing fretting was indicated by the final two failures, which did not have fretting origins. All step-test failures were at flexing angles larger than ±5° with 30,000 lb-in. of torque, and after lives longer than 10 million cycles. The step-test without failure accumulated over 90 million cycles. The flexing angles were not proven equal to a rotating driveshaft misalignment angle. Rotating driveshaft tests were expected to establish the relationship between the test flexing angle and driveshaft misalignment.

Additional useful tests were accomplished. After each failure, a test to second failure was run at reduced torque and various angles to explore fail-safe coupling life. At 6000 rpm, the test coupling life to second failure would be approximately twenty minutes at 16,000 pound-inches torque and a misalignment angle equivalent to the test flexing angle of ±4°. Test couplings were loaded to 30,000 pound-inches torque after two failures to demonstrate a remaining torque-carrying capability. A standard salt spray test found non-embrittling cadmium plate and nickel cadmium plate acceptable for environmental protection of maraging steel while electroless nickel plate was not acceptable. Bolt pre-load retention was investigated with no observed loss of bolt pre-load and no observed bolt yielding or creep. An ultimate torque test performed on an aluminum version of the test coupling predicted ultimate torque capacity of the test coupling to be 75,000 pound-inches.

Program Purpose

The purpose of this program is to perform experimental flight testing of the Bossler coupling in an HH-2D helicopter.

Program

This program included test driveshaft definition; bench tests of a rotating driveshaft assembly using Bossler couplings under combined speed, torque and misalignment; an experimental flight test program; and identification of concepts for increasing capability.

TEST DRIVESHAFT DEFINITION

Introduction

This section discusses the selected application, lists the design requirements and describes the test specimen.

Design Application

It was desired that the test specimen meet the requirements of a significant and current application, which would eventually permit a direct comparison of capabilities with an existing successful design. Therefore, a test driveshaft assembly was designed as a direct replacement for the standard driveshaft currently used in HH-2C and HH/SH-2D helicopters to connect the engine combining transmission to the main transmission.

The standard driveshaft uses two gear-type couplings. It transmits 18,000 lb-in. of torque at 6000 rpm. The static installed misalignment is 0.5°. The operating angular misalignment is 1° maximum. The operating axial change-of-length is ±0.15 inches maximum. Each coupling allows 4° misalignment while the driveshaft assembly is axially compressed or extended 0.30 inches. A disconnect design is provided so that either the main transmission or the engine combining transmission can be removed without disturbing the one not removed. Also, the driveshaft assembly can be installed and removed without disturbing either transmission. The lowest lateral natural frequency is more than 1.15 times the maximum operating speed. The gear couplings signal distress by overheating prior to failure, which is detectable by periodic inspection of temperature tapes mounted on the couplings. The tapes change from white to black when the coupling exceeds the allowable operating temperature.

Design Requirements

Criteria were provided for design and test of the experimental driveshaft assembly. The objectives were increased reliability, longer life and the elimination of periodic maintenance. The criteria are listed below:

(1) Torque Capacity
Continuous Torque = 18,000 pound-inches
Yield Torque = 37,900 pound-inches
Ultimate Torque = 54,000 pound-inches

(2) Angular Misalignment
Continuous 2°
Transient 4°

(3) Axial Deflection
Each coupling $\pm$.125 inches, driveshaft assembly $\pm$.25 inches simultaneously with all other conditions.

(4) Speed
Operating speed is 6000 rpm
The lowest lateral natural frequency shall be more than 7000 rpm.

(5) Maximum End Reaction
Reaction $\leq \pm 600$ pound-inches end moment

Reaction $\leq \pm 300$ pounds axial force

Reaction $\leq$ 200 pounds radial force

Reaction $\leq$ 0.10 ounce-inches unbalance at 6000 rpm

(6) Environmental Protection
Corrosion protection per applicable Mil Specs.
Permanent fasteners made tamper proof.
Temperature Range from -65°F to 180°F.

(7) Installation
Interchangeable with present driveshaft assembly including length of 23.50 inches and a target weight of 17 pounds.

Procedure suitable for line-level maintenance personnel.

No new tools.

The driveshaft assembly shall have a shear disconnect design so that either gearbox can be removed without disturbing the other and so that the driveshaft assembly can be removed without disturbing either gearbox.

(8) Life
Design life shall be 10,000 hours without maintenance.

(9) Fail-Safe
The couplings shall have a fail-safe design that retains all parts, allows torque transmission to continue and signals the failure.

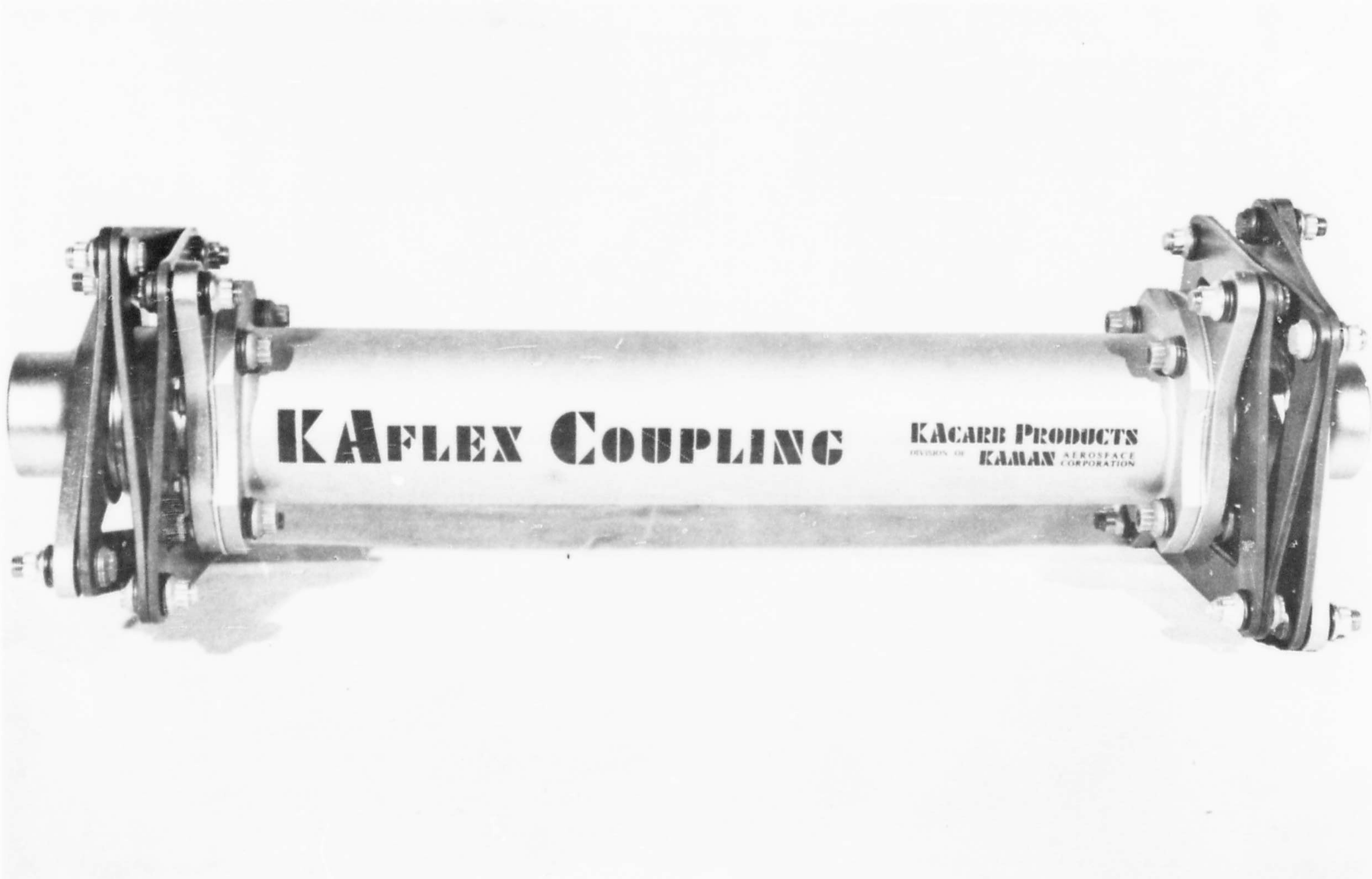


Figure 1. Driveshaft Assembly

Description of Test Driveshaft

The driveshaft assembly is shown in Figure 1. The component parts are identified on the schematic sketch, below.

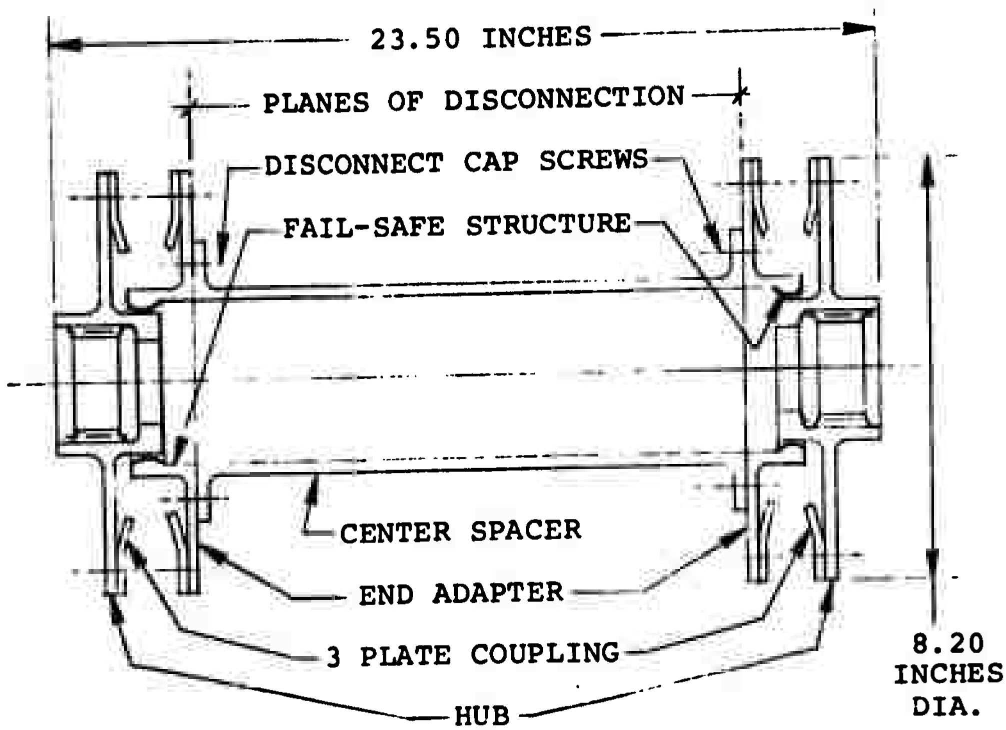


Figure 2. Schematic, Driveshaft Assembly

The lenqth is 23.50 inhces, the maximum diameter is 8.20 inches. The assembly consists of 2 splined hubs, 2 three-plate couplings, 2 shaft end adapters, 1 center spacer and attaching hardware.

Fail-safe design for the couplings is provided as specified in the Design Requirements. Fail-safe structure is incorporated in the hub and end adapters. The structure consists of short, concentric, overlapping shafts which mechanically entrap the center spacer. The radial clearance between the

overlapping shafts is small. Any coupling failure causes the center shaft to move laterally until the overlapping shafts limit further lateral motion. Torque transmission will continue through the surviving coupling elements. An axial force on the center spacer is developed by this torque transmission. The axial movement of the center spacer is also limited by the fail-safe structure. The lateral and axial movement of the center spacer cause an unbalance which, in turn, causes a vibration to signal the failure.

Shear disconnect design is provided as specified in the Design Requirements. Shear disconnect disassembly is accomplished by removing the cap screws and then the center spacer from the adjacent end adapters.

The hub, coupling and end adapters are to be permanently joined together. The bolts used in the permanent assemblies have their heads shortened to increase internal coupling clearance. The nuts used in the permanent assemblies may be riveted to the bolts to prevent disassembly.

All design requirements were met except the target weight. The test driveshaft weight is 20.5 pounds, which exceeds the target weight of 17 pounds. The design shown includes features useful for test, inspection and convenience of modification that would be eliminated in a production design. It is estimated that the target weight of 17 pounds could be met. The detail design of the test assembly is illustrated in Figure 3 - Driveshaft Cross-Section, Full Scale.

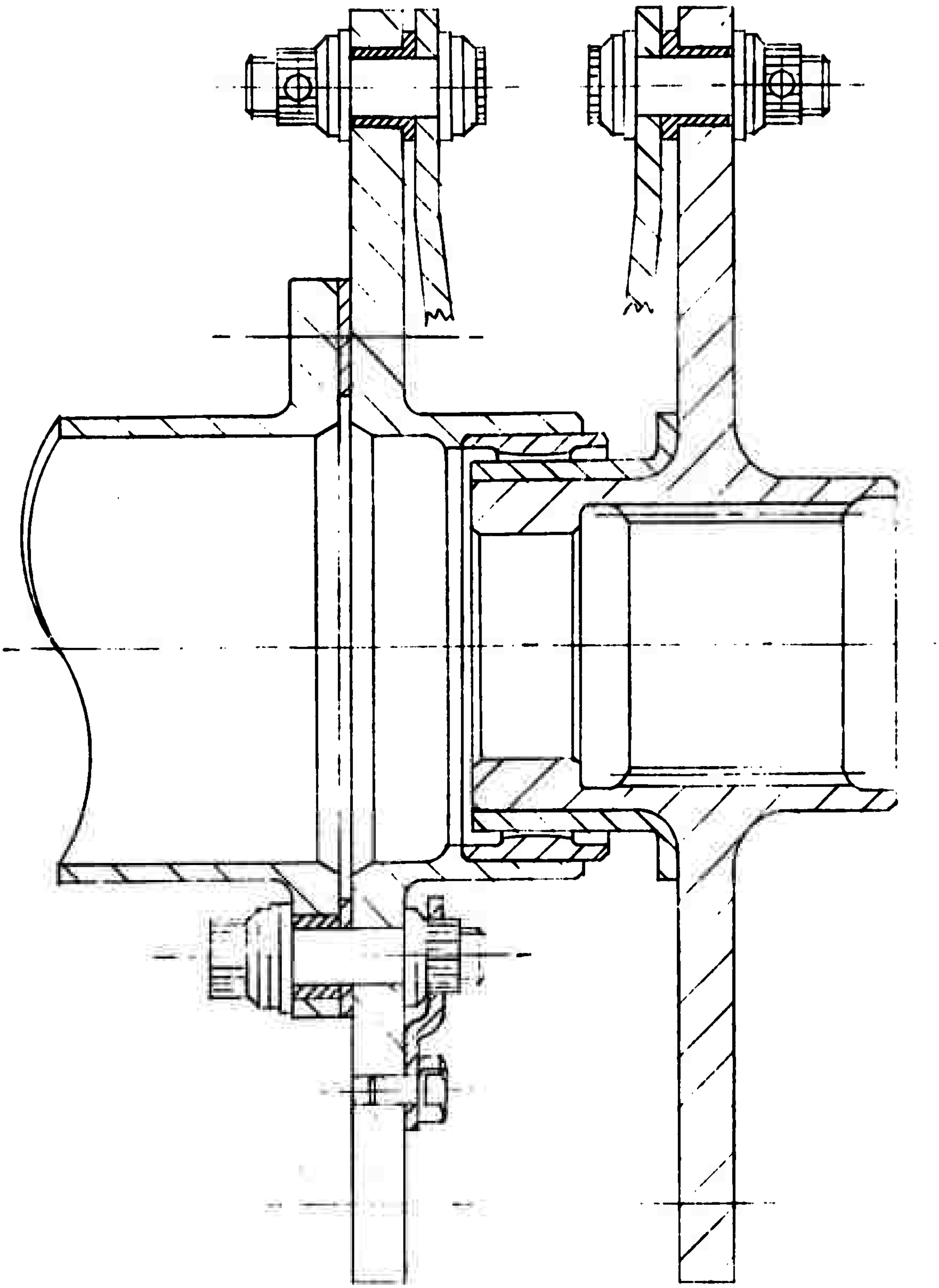


Figure 3. Driveshaft Cross-Section, Full Scale

BENCH TESTS

Introduction

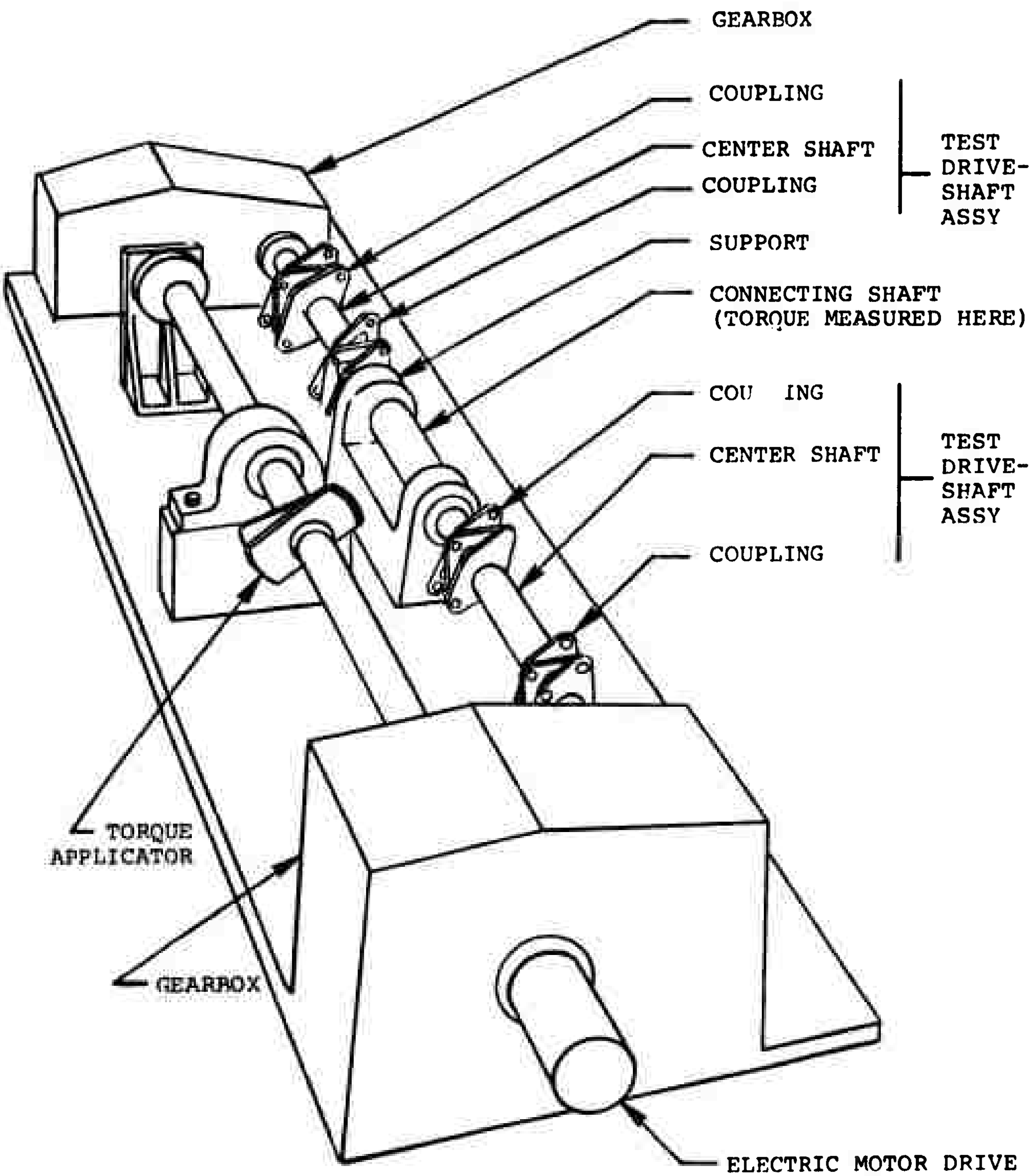
The purpose of the bench tests is to reduce the possibility of a driveshaft failure during service. As such, the bench tests must reproduce service conditions as accurately as possible. Many samples should be tested. Failures must occur. The stress levels at which failures occur must be sufficiently removed from service conditions so that confidence in survival is generated. The service conditions which were applied to full-scale driveshaft assemblies by the bench tests were speed, torque and misalignment. It was found that failure during flight test was extremely unlikely.

Apparatus

Driveshaft assemblies were endurance tested on the bench test rig shown on Figures 4, 5, and 6. The rig is a regenerative type. It was used originally for qualification testing of the UH-2 driveshaft. The rig was extensively modified for the Bossler coupling experimental flight test program. Torque capacity was increased to 18,000 pound-inches. Provision was made to test two driveshafts simultaneously by shortening the connecting shaft and adding a support bearing. The connecting shaft support structure is adjustable in location. Thus, each of four couplings can be tested simultaneously at four different angles. The speed at which the specimens were tested is determined by electric motor speed and the gearboxes; it is 6200 rpm. Torque is applied to the low-speed shaft and measured by strain-gages on the connecting shaft between the test driveshaft assemblies. Vibration is measured by accelerometers which were located at various places during testing. Other test rig instrumentation includes oil temperatures and pressure.

Test

Ten different couplings on two different driveshaft assemblies were tested prior to release of the design for flight evaluation. All tests were run at 6200 rpm and 18,000 pound-inches of torque. Misalignment was increased to levels which produced two coupling failures and nine coupling survivals of long duration.



DRIVESHAFT TEST RIG

Figure 4

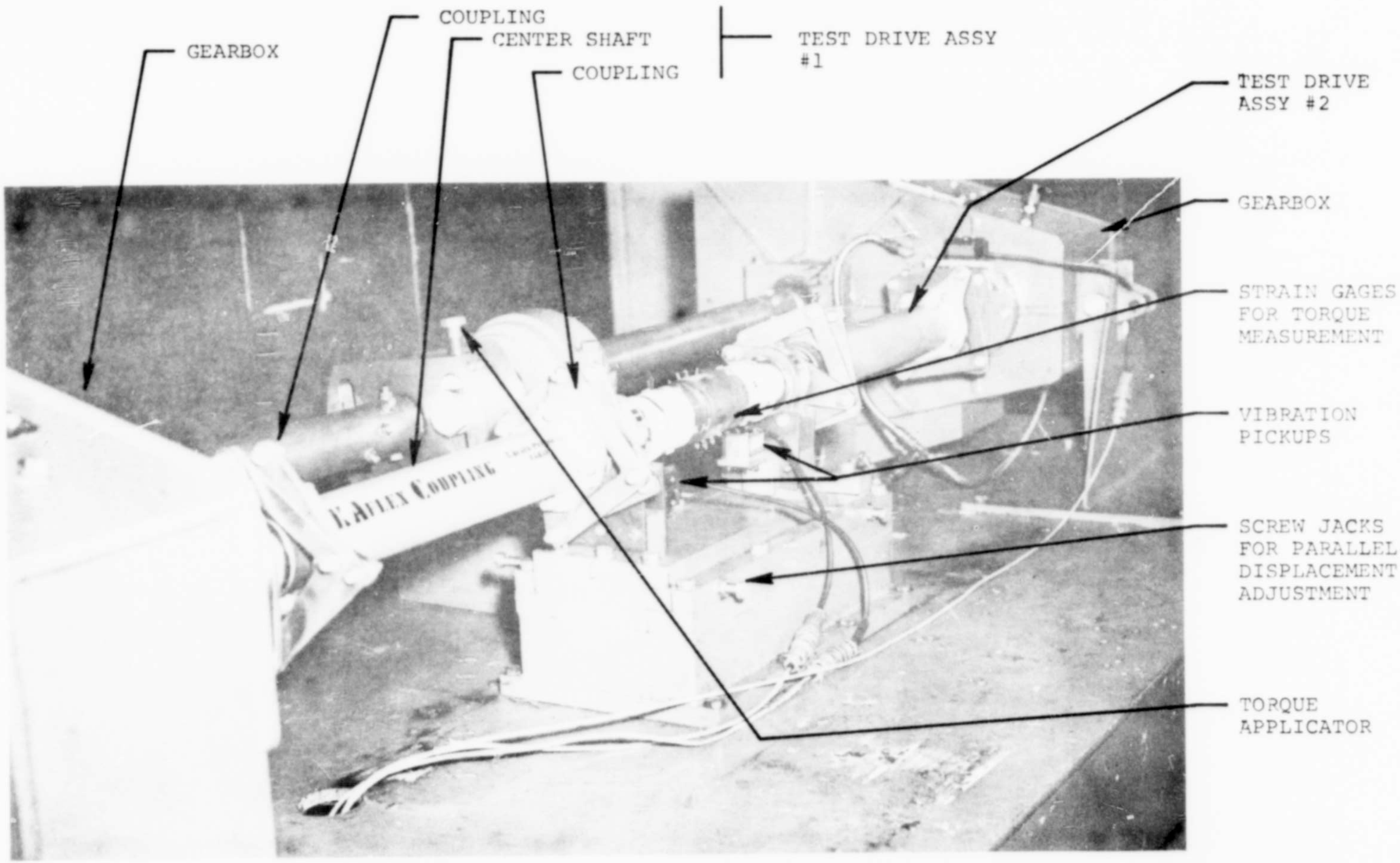


Figure 5. Perspective View of Test Rig

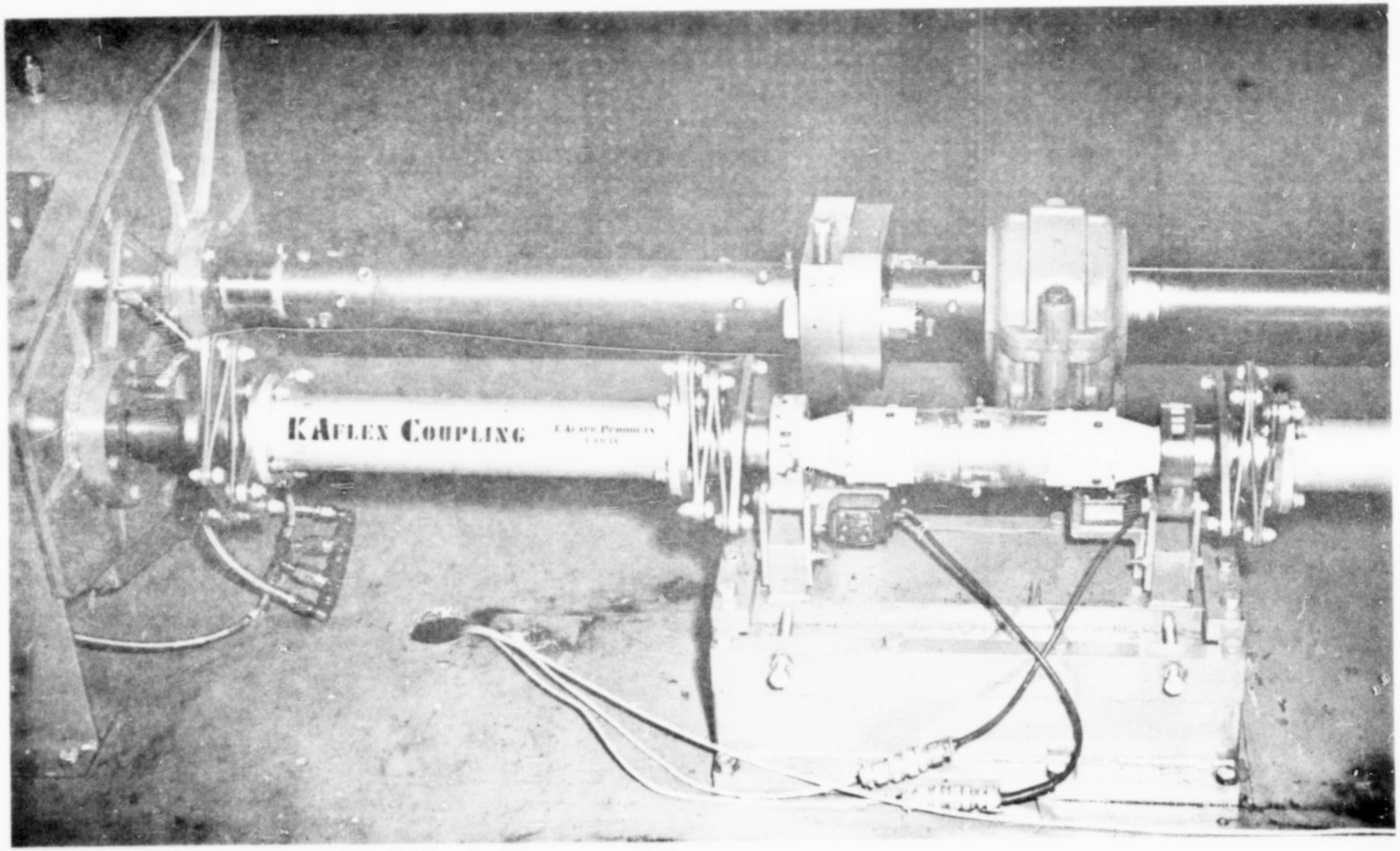


Figure 6. Elevation View of Test Rig

The term "misalignment angle" is used for convenience. Actually, the input and output hubs of each driveshaft are essentially parallel but are displaced laterally. Each coupling is then a short cantilevered beam fixed to each hub. The greatest bending moment on a cantilever beam occurs where it is fixed to structure. Therefore, the three plates which comprise the coupling do not share the misalignment equally. The plates attached to the input and output hubs have the highest bending moment, therefore, the greatest angular deflection and the greatest alternating stress. The non-uniformity of stress distribution depends on the length of the center shaft and the length of the coupling. In this design, the angle between adjacent shaft centerlines is a reasonably accurate approximation of the true angle to which the end plate was exposed. As was to be expected, however, all failures occurred in the end plate attached to the hub at the coupling location having the largest misalignment angle of the four coupling locations.

Figure 7 shows the test misalignment angles that were applied to each of the four coupling locations. The misalignment angle is the angle between adjacent shaft centerlines. Table I indicates the location of each of the ten test couplings during endurance testing. These ten couplings and the two driveshaft assemblies accumulated 196 million cycles in 528 hours of rig operation. The testing was accomplished by four separate endurance runs. The history of each endurance run is summarized below.

<u>Run Number One</u>

Test Coupling Number	Misalignment Angle	Time	Cycles
1	2.10°	16 hrs. 22 min.	6.09×10^6
2	1.99°	16 hrs. 22 min.	6.09×10^6
3	2.49°	16 hrs. 22 min.	6.09×10^6
4	2.635°	16 hrs. 22 min.	6.09×10^6

Testing prior to Run Number One included 5 min. at 0°, 10 Hrs. at 3/4°, 29 hrs. 39 min. at 1-1/2°, 1 hr. at 2°. The test rig was then adjusted to the test angles listed, which were used for Run Number One and all subsequent tests. A bolt

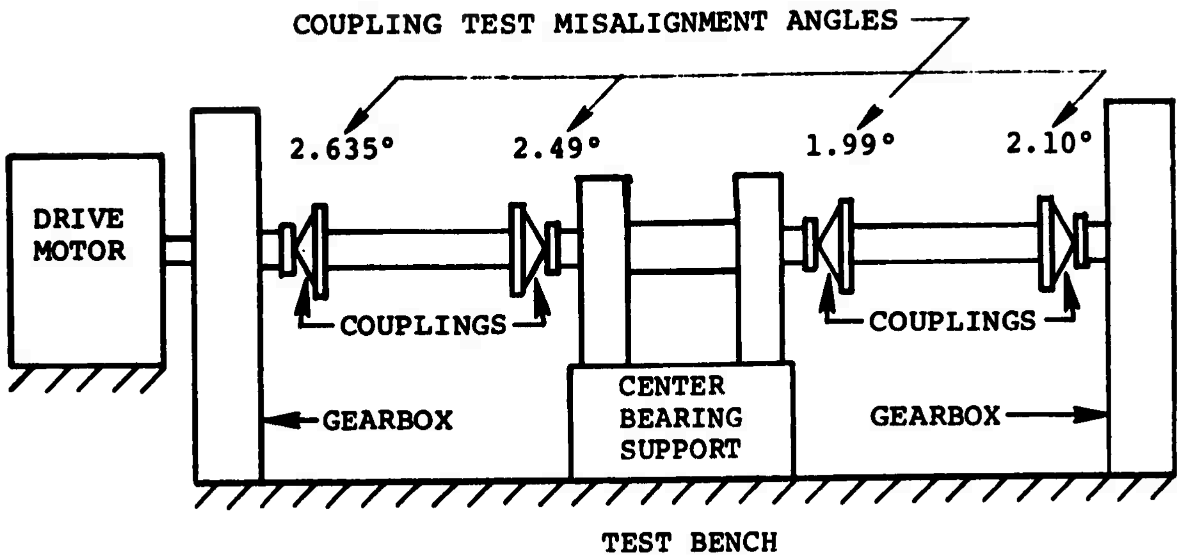


Figure 7. Coupling Test Misalignment Angles

TABLE I. TEST LOCATION OF TEN COUPLINGS				
Test Run No.	Test Coupling Numbers			
One	4(2.635°)	3(2.49°)	2(1.99°)	1(2.10°)
Two	8 "	7 "	6 "	5 "
Two (Cont)	8A "	7 "	6 "	5 "
Three & Four	10 "	9 "	6 "	5 "

failure occurred at Coupling No. 4 at 12 hrs. 15 min. It was found that the bolt had been scored by the hardened washer which was clamped between the end plate and the hub. These washers were replaced by a hub-mounted bushing which prevented any subsequent bolt damage.

A plate failure occurred at Coupling No. 4 at 16 hrs. 22 min., concluding Run Number One. The failure was a typical fatigue failure with the origin in the contact area where the end plate is clamped against the face of the bushing in the hub. The fail-safe retained all parts and allowed torque transmission to continue. Total time at all angles was 57 hrs. 6 min. The driveshafts were disassembled and all parts were inspected and magnafluxed.

Run Number Two

Test Coupling Number	Misalignment Angle	Time	Cycles
5	2.10°	27 hrs. 30 min.	10.2×10^6
6	1.99°	27 hrs. 30 min.	10.2×10^6
7	2.49°	27 hrs. 30 min.	10.2×10^6
8	2.635°	10 hrs. 55 min.	4.06×10^6
8A	2.635°	16 hrs. 30 min.	6.13×10^6

A plate failure occurred at No. 8 Coupling at 10 hrs. 55 min. The failure was a typical fatigue failure with the origin in the contact area where the end plate is clamped against the face of the bushing in the hub. The fail-safe retained all parts and allowed torque transmission to continue. The broken plate was replaced by a plate from Coupling No. 2. Testing continued until Couplings 5, 6, and 7 reached 10 million cycles and Coupling No. 8A had 6.13 million cycles. This concluded Run No. 2. Driveshafts were disassembled. All parts were inspected and magnafluxed.

Run Number Three and Four

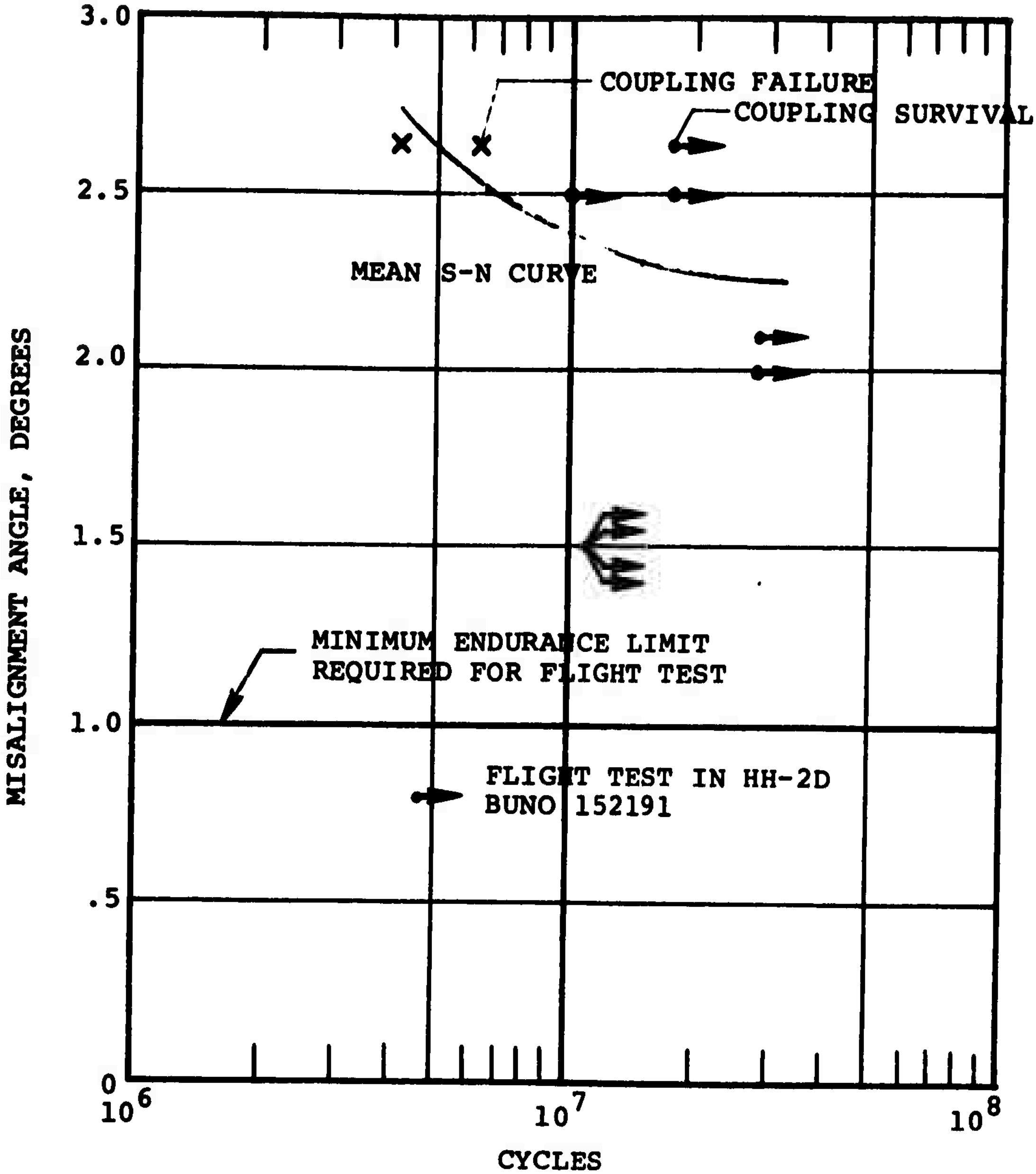
Test Coupling Number	Misalignment Angle	Run #2 Hrs	Run #3 Hrs	Run #4 Hrs	Total Time Hrs	Total Cycles
5	2.10°	27.5	27	20.5	75	27.9×10^6
6	1.99°	27.5	27	20.5	75	27.9×10^6
9	2.49°	-	27	20.5	47.5	17.65×10^6
10	2.635°	-	27	20.5	47.5	17.65×10^6

Plates used on No. 9 and No. 10 Couplings were taken from Couplings Numbers 1, 2, 3, and 4, which had a total prior time at all angles of 57 hrs. 6 min. Run #3 concluded after 27 hours without failure, which completed contract requirements for flight test.

It was decided to try for 30×10^6 cycles on Couplings Numbers 5 and 6. Parts were disassembled, inspected, magnafluxed, and reinstalled. Run No. 4 concluded after 20.5 hrs. due to test rig failure. The total at all angles on Couplings No. 9 and No. 10 is 104 hrs. 36 min. Driveshafts were disassembled, parts inspected and magnafluxed. No discrepancies or indications of degradation which might lead to failure were found.

Results and Discussion

The significant test data are plotted on Figure 8 and are used to locate a tentative mean S-N curve. By convention, the symbol "S" stands for alternating stress while "N" represents number of cycles. The significance of the mean S-N curve is that half the couplings are expected to survive the alternating stress for the number of cycles defined by any point on the mean S-N curve. Failure values often show considerable scatter above and below the mean S-N curve. The ordinate is scaled in misalignment angle (angle between adjacent shaft centerlines) rather than alternating stress because misalignment angle was the measured control of alternating stress. Because the S-N curve for steel is virtually horizontal for cycles larger than 10^7, a test specimen that endures 10^7 cycles or more without failure is expected to have infinite life and is called a runout. Survivals with less



NOTES: (1) All tests were run at 18,000 lb-in. torque and 6200 rpm.

(2) Bench test survivals with less than 10^7 cycles are not shown.

(3) 10^7 cycles required 27 hours.

Figure 8. Test Data and Flight Envelope

than 10^7 cycles are not considered to be significant and are not shown on Figure 8. Runout test points are shown with an arrow to indicate no failure when the test was terminated. Figure 8 also shows the flight envelope and the flight test data point. The bench testing was finally terminated due to a rig failure not associated with the coupling.

It was observed that all the coupling plate failures originated in the contact areas where the end plate is bolted against the bushing face of a shaft hub, see Figures 1, 2, and 3.

The contact area suffered a stress concentration due to fretting in an area of alternating bending stress. Investigation of this problem area in the previous contract program had led to treatment for suppression of fretting damage. However, further study suggested that the bending stress could be reduced as well. Briefly, the plate thickness would be increased in the contact area of the coupling plate, with a compensating thinning in regions where stress concentrations do not occur.

Conclusions

1. All coupling fatigue failures originated in the contact area of a coupling end plate on the bushing face of a hub. The local bending stress in the contact area is aggravated by stress concentration due to fretting. The local bending stress could be reduced by increasing plate thickness in the contact area and decreasing plate thickness elsewhere.

2. After a coupling failure during bench testing, the fail-safe retained all parts, allowed torque transmission to continue and signalled the failure by an increase in vibration level.

3. Driveshaft behavior and life after a coupling failure in a helicopter is not known, because torque was reduced on the bench test by a coupling failure, and driveshaft rotation was stopped as soon as a failure was observed.

4. The minimum endurance limit required for flight test is less than half the tentative mean endurance found by bench test, so that failure during a limited flight test of less than $.5 \times 10^7$ cycles is extremely unlikely.

FLIGHT TEST

Introduction

The main driveshaft in the UH-2D helicopter is an existing successful design in a significant current application. The existing driveshaft provides a good base for comparison. The UH-2 application provides the opportunity for observing compatibility with a known demanding environment.

Installation

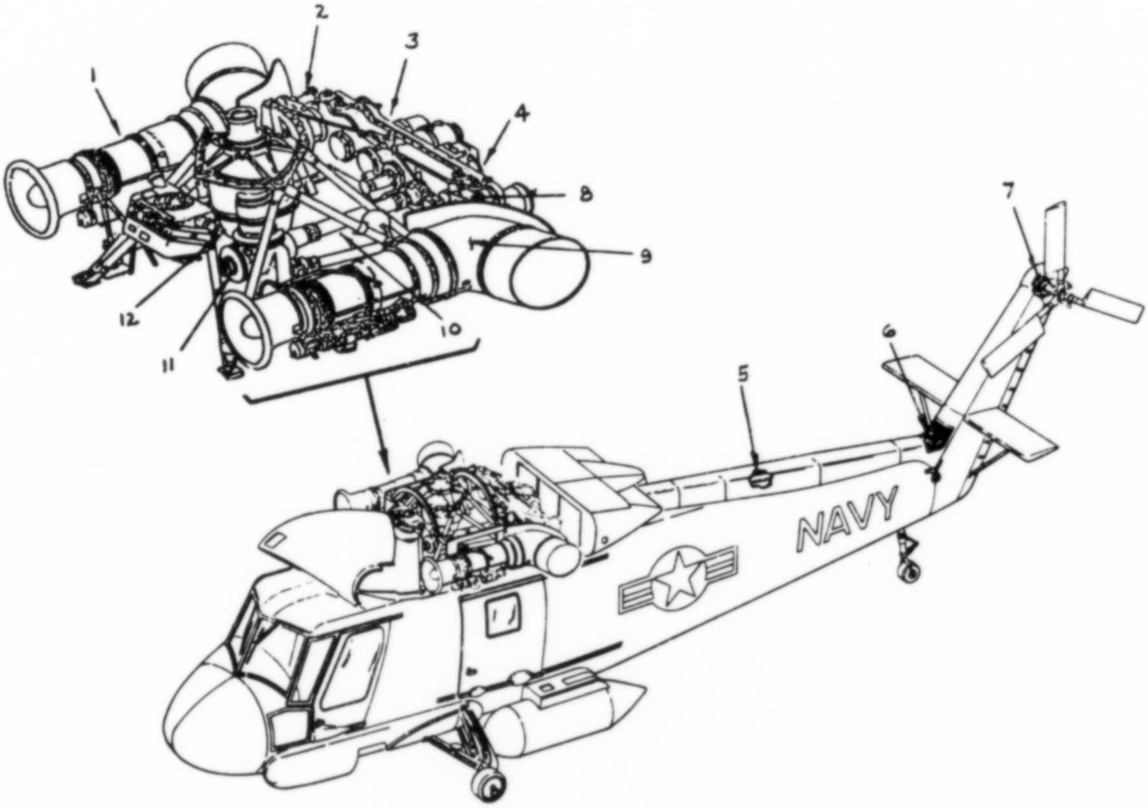
The powerplant and drive arrangement of the UH-2C and HH-2D helicopter is shown in Figure 9. The main driveshaft is Item 10 on Figure 9. The transmission system is shown on Figure 10, with the main driveshaft identified as Item 5. Figure 11 is a photograph of the test driveshaft installed in HH-2D BuNo 152191.

The gearbox housings adjacent to the driveshaft end connections were instrumented to measure vibration and stress. The combining gearbox accelerometers measured vertical, horizontal, and axial vibration. The main transmission accelerometers measured vertical and horizontal vibrations. Strain gages were attached to both gearboxes. A potentiometer driven by a rod parallel to the test driveshaft measured axial change-of-length. Rotor torque and torque of each engine was measured.

Test

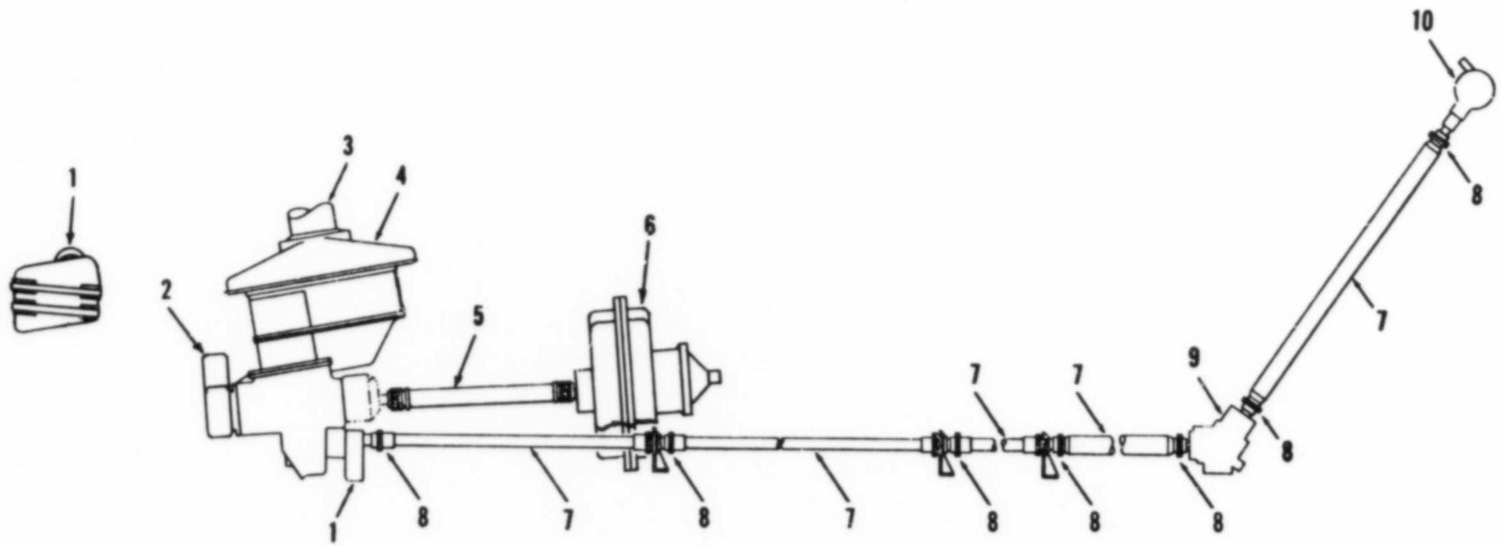
A vibration/strain/displacement survey was made using the existing PD-1320 production driveshaft. The HH-2D flight envelope was flown with the aircraft at 11,200 pounds gross weight. The test driveshaft was then installed and natural frequencies measured. During flight test, records of gearbox vibration, strain, and displacement were taken and inspected in order to detect if any undesirable effects were caused by the test driveshaft. No questionable driveshaft behavior was observed.

The flight test consisted of one hour ground run and taxi, one hour hover, one hour hover and maneuver in ground effect, followed by a flight envelope which included autorotation from 70, 100, and 120 knots entry speed, V_{max} of 142 knots, jump take-off, turns and flares. The peak torque was 110% of design at 100% rpm.



1. No. 2 engine
2. No. 2 engine speed decreaser gearbox
3. Combining gearbox
4. Rotor brake
5. Tail rotor drive shaft
6. Intermediate gearbox
7. Tail rotor gear box
8. No. 1 engine speed decreaser gearbox
9. No. 1 engine
10. Main drive shaft
11. Transmission oil pump
12. Main gearbox

Figure 9. Power Plant and Drive Arrangement, UH-2C, D Helicopter



1. *Oil supply system*
2. *Resolver and tachometer drive gearbox*
3. *Main rotor shaft*
4. *Main gearbox*
5. *Main drive shaft*
6. *Combining gearbox*
7. *Tail rotor drive tubular shaft*
8. *Tail rotor drive coupling*
9. *Intermediate gearbox*
10. *Tail rotor gearbox*

Figure 10. Transmission System

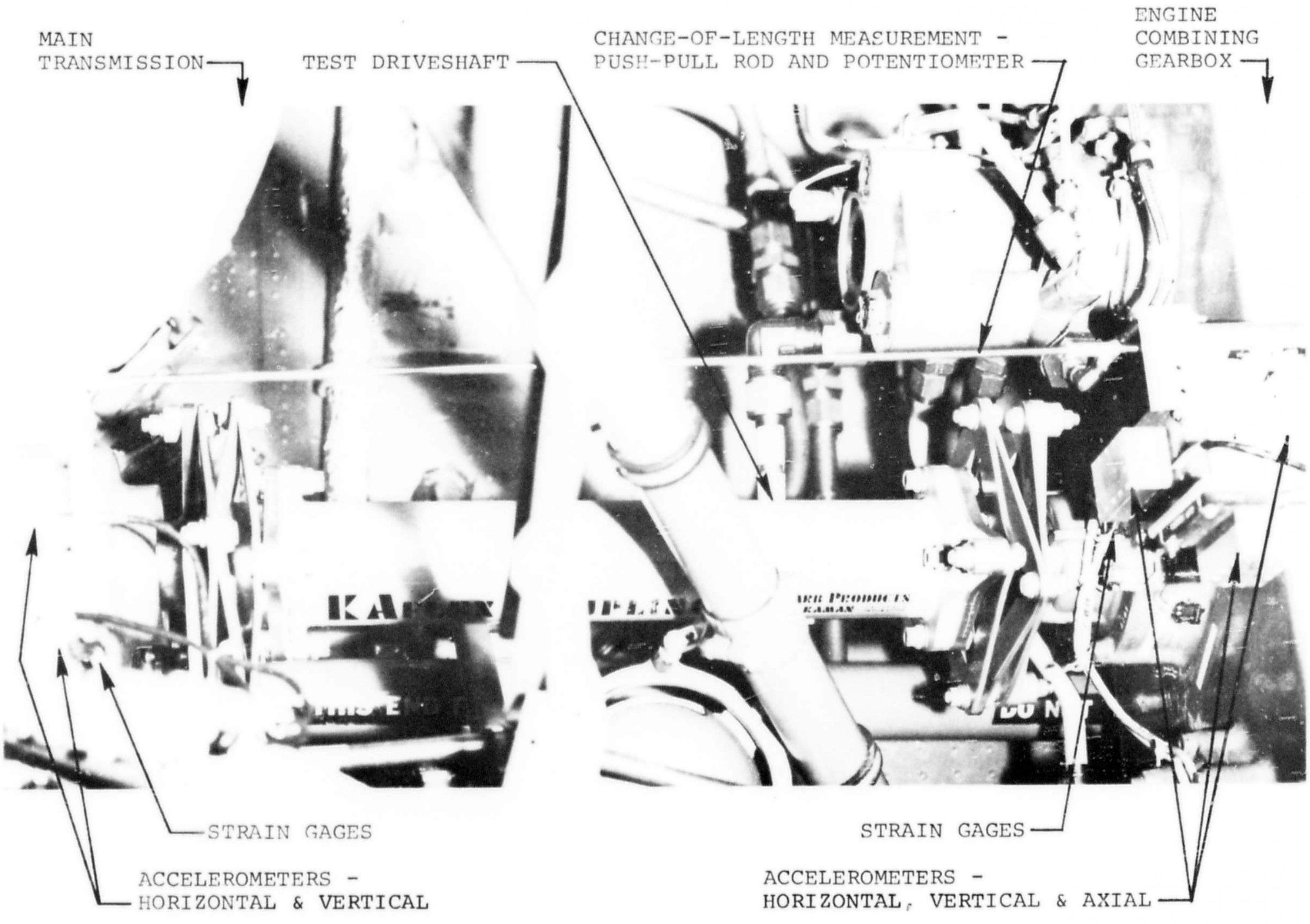


Figure 11. Test Driveshaft Installed in HH-2D

The last four hours were flown at an overload gross weight of 12,600 lbs. Design gross weight is 12,500 lbs. Essentially the same flight envelope was flown, except that V_{max} was 136 knots. No questionable driveshaft behavior was observed. A follow-up flight was then conducted with the standard driveshaft at the 12,600 lb gross weight to furnish a comparison with the test driveshaft at the 12,600 lb gross weight.

The total flight test elapsed time using the test driveshaft was 12.5 hours.

Results and Conclusions

The test driveshaft behavior in the helicopter environment and the subsequent test driveshaft disassembly, inspection and review all indicate that the test driveshaft is fully capable of further use. It was concluded from the bench and flight tests that up to fifty hours flight testing under engineering surveillance and control could be accomplished safely and to useful purpose with the test driveshaft. Permission for this continued flight test was requested by Kaman and granted by the Naval Air Systems Command.

CONCEPTS FOR INCREASING CAPABILITY

Introduction

The fundamental goal of this research program was to perform experimental flight testing of the Bossler coupling in an HH-2D helicopter in order to establish feasibility. The purpose was successfully accomplished. Further, it was found that continued flight test is desirable. This has been agreed to by Kaman and the Navy. Additionally, the investigation identified means of increasing further the Bossler coupling's capability. Two such concepts are discussed briefly below.

Redistribution of Bending Stresses

As noted previously, the two coupling fatigue failures which occurred at the highest misalignment angles originated in the contact area of a plate on the bushing face of the hub to which the plate is bolted. It appears possible to reduce the bending stress in the contact area where fretting provides a stress concentration factor. The local bending stress can be reduced by increasing plate thickness in the contact area and decreasing plate thickness elsewhere. Thus, flexure (and bending stress) would decrease in an area of known stress concentration and increase in areas where the stress concentration factor is lower, for a more uniform product of stress and concentration factor. Increased misalignment capability for the same weight will result from the more efficient use of material.

This design modification is considered to be significant for future inquiry. Design and test of couplings suitable for the main driveshaft reported herein is recommended. Such a program would allow a direct comparison of improved misalignment capability as well as an improvement in the test driveshaft.

Operation Above Rigid Body Resonant Speeds

As noted in the Introduction Section of this report, proportioning the flexing members of a flexible coupling is a trade-off between static capacity, stiffness, and misalignment capability. Strong thick stiff members easily transmit large torques and provide high natural frequencies. However, misalignment requires flexing of these members. The flexing produces alternating stress that can limit coupling life. The greater the static strength and stiffness of a flexing member, the higher the alternating stress from a given misalignment. Therefore, strength and stiffness provisions to transmit torque at speed will be detrimental to misalignment capability.

The criteria to which the present test driveshaft was designed included a stiffness requirement that caused plate thickness to be increased above that necessary for safe torque transmission, increasing weight and decreasing misalignment capability. The specific requirement was that the lowest transverse natural frequency be more than 115% of the highest operating speed.

The center shaft supported by Bossler couplings behaves as a spring-mass system. The lowest natural frequencies are rigid body resonant speeds, with a large speed difference between the rigid body resonant speeds and the lowest bending resonant speed. Operation in the speed range between the rigid body resonant speeds and the lowest bending resonant speed permits the use of more flexible coupling plates, thereby providing a high-speed high-angle coupling. Many successful designs of rotating machinery are known, including two production helicopter driveshafts, in which the shaft speed is higher than the lowest transverse natural frequency. In both the helicopter cases, the driveshaft is restrained from large motions as it passes through the resonant speed by special restraining structure.

The fail-safe design evolved for use with the Bossler coupling provides such a restraining structure to allow passage through resonance. The restraining force is the centrifugal force produced as the restrained weight of the vibrating hardware passes the resonant speed during acceleration or deceleration. Because centrifugal force varies as the square of the speed, it would be desirable to pass through resonance at as low a speed as possible. Thus, the couplings could be designed to be as flexible as torque transmission requirements allow in order to reduce the resonant speeds as much as possible. The bias toward increasing flexibility would reduce coupling weight, end-moment and axial force for a design misalignment at a design speed or, conversely, misalignment and speed could be increased. The latter is especially important because of the increasing requirements of modern transmission technology for lighter structures with greater displacements and ever increasing speeds. The amount of weight that can be saved, or the increase in misalignment that can be endured, are a function of the torque, speed, length and weight of the driveshaft application. It is estimated that the test driveshaft for this program, if designed to operate above the rigid body resonant speeds, would have a weight reduction of between two and three pounds, an

increase in allowable misalignment on the order of $\pm 1°$, and a maximum allowable operating speed at least four times as high as the present design.

The identification of this area of increased potential capability is considered to be significant for future inquiry. Design and test of a driveshaft assembly using Bossler couplings which operate above the rigid body resonant speeds is recommended. The design application should be one where the potential gains are large and significant.

CONCLUSIONS

1. All design requirements except the target weight were demonstrated, including a long endurance life at the design flight conditions.

2. It may be possible to increase coupling capability for high-angle high-speed applications by designing to operate above the rigid body resonant speeds, using the fail-safe to limit center section displacement during transition.

3. All coupling fatigue failures originated in the contact area of a coupling plate on the bushing face of the hub to which the plate is bolted. The local bending stress in the contact area is aggravated by stress concentration due to fretting. The local bending stress could be reduced by increasing plate thickness in the contact area and decreasing plate thickness elsewhere.

4. After a coupling failure during bench testing, the fail-safe retained all parts, allowed torque transmission to continue and indicated the failure by an increase in vibration level.

5. Driveshaft behavior and life after a coupling failure in a helicopter is not known, because torque on the bench test was reduced by a coupling failure, and driveshaft rotation was stopped as soon as a failure was observed.

6. The tentative mean endurance found by bench test is more than double the minimum endurance limit required for flight test, so that failure during a limited flight test is extremely unlikely.

7. The driveshaft assembly used in the flight test program is fully capable of further flight use.

8. The bench and flight test results justify up to 50 hours of experimental flight test under engineering surveillance and control.

RECOMMENDATIONS

1. Continue experimental flight test up to fifty hours under engineering surveillance and control.

2. Investigate reducing bending stress via local plate thickening in the contact areas where all fatigue failures originated. Simultaneously investigate local plate thinning in areas where flexure is preferred.

3. Investigate a high-angle high-speed coupling designed for driveshaft operation at a speed below the lowest bending frequency but above the rigid body modes, using the fail-safe to limit center section displacement during transition.

4. Continue investigation of the effectiveness of the fail-safe provisions by bench testing under specially simulated failure conditions to establish residual torque capacity and safe operating interval.

5. Continue endurance testing of representative design couplings to enlarge the statistical base for safe life prediction.

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